Bolts and Bolted Members

By JAY ANTENEN, M.E. IV

MANY senior machine design students were rudely awakened last fall by "Pop" Norman's startling comment that "any bolt under ½ inch in diameter was unsafe and should never be used." As all of us had encountered an astronomic number of bolts this size holding things together, this seemed an odd statement, especially when couched in the Swedish accent of "Pop" Norman. However, it is absolutely true that if a bolt ½ inch or under in diameter is drawn up tightly to resist a load, the initial stresses involved are of such nature that the bolt has no strength left. To prove it, the good professor proceeded to relate some tests made by some ingenious Cornell men, Kimball and Barr, in the 1920's. These fellows it seems, called in a dozen experienced mechanics, gave them a choice of wrenches, and instructed them to screw up a bolt and nut, placed in a testing machine so the load on the bolt could be weighed, to what they thought would make a steam tight joint. The results showed, among other things, that an average mechanic must be a hard thing to find because the amount of initial load put on the bolt varied all over the place. They did indicate, however, that (a) the initial load due to screwing up varied as the diameter of the bolt and (b) the initial load due to tightening was about 16,000 lb. per inch of bolt diameter.

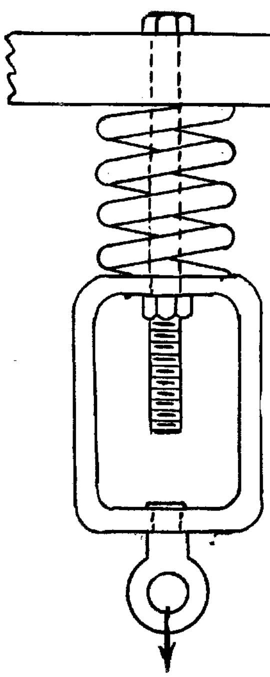

Fig. 1

Rigid bolt and elastic members

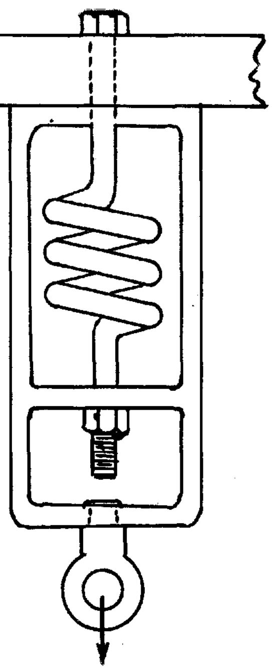

Fig. 2

Elastic bolt and rigid members

The intensity of the load can be obtained by dividing the load by the area at the bottom of the threads:

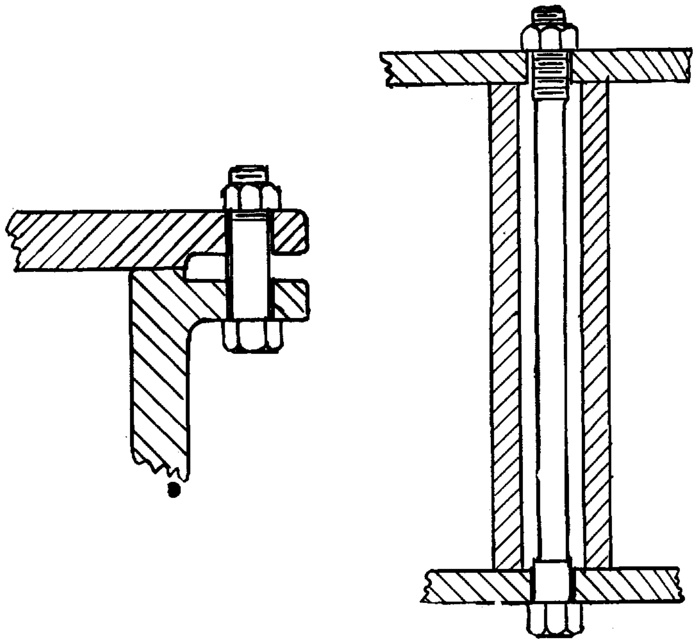

Fig. 3. Practical cases that approach the two limiting conditions of rigidity and elasticity

$$S = \frac{P}{Ar} = \frac{16{,}000d}{\frac{\pi d^2}{4}} = \frac{27{,}500}{d} \text{ psi}$$

If d is ½ inch, $\frac{27{,}500}{\frac{1}{2}} = 55{,}000$ psi which is above the ultimate tensile strength of many bolt steels. Some of the heftier mechanics managed to shear off ½ inch bolts in tightening them in the Cornell tests. The fact that there is is an initial torsional load incurred when the nut is tightened is taken into account in a more profound equation, $P = S_t\,(0.55d^2 - 0.25d)$, which Forrest E. Cardullo brought out a little later. This equation also show that a ½ inch bolt has no strength at all.

So far, the stresses mentioned are initial stresses due only to the screwing up of the nut. In bolts subject to alternating external loads the factor of initial tightness assumes a major importance, accounting for about 90 percent of the strength of the bolts. What will be the effect of the initial tension when combined with an external useful load? Will the resultant load be the sum of the initial and external loads or will the addition of an external load cause no addition to the initial stress unless it exceeds the initial load? It happens that both these views represent the extreme limiting cases and every real proposition will lie in between them somewhere. These cases are (1) when the bolt is absolutely rigid and the bolted members yield under the pressure and (2) when the bolt alone is yielding and not the members.

The first of these limiting conditions is approached by Fig. 1. If the nut is screwed up until

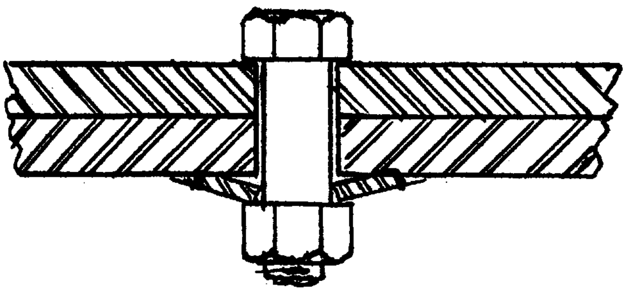

Fig. 4—An elastic washer will add to the elasticity of the bolt.

the compressive force on the nut is 1,000 lb., the load on the bolt will be 1,000 lb. The bolt can be considered rigid as its modulus of elasticity is very much removed from that of the spring. Now, if an external load is applied to the yoke, the spring will exert the same force as before and the bolt will have to support both the external and the bolted load. In any possible case, the resultant load is somewhat less because the bolt can not be made absolutely non-yielding.

The other limiting case is approached by the set up shown in Fig. 2. In this case we have the ideal elastic nut as compared to the bolted yoke member. If an initial load of 1,000 lb. is put on the bolt, the yoke will be drawn up with a force of 1,000 lb. against the support. Now again if an external load is applied to the yoke, a load of 1,000 lb. is required to pull it away. Therefore, the external load does not affect the initial load until it becomes larger, and then it alone is responsible for the stress in the bolt.

In all practical cases, the difference in elasticity between the bolted members and the bolt is nowhere near these springs and bolts, but in a particular case, the engineer should be able to determine which limit is the more nearly approached and govern his design accordingly. In practice, the situation of Fig. 3 somewhat approximates the limiting cases.

The American Institute of Bolt, Nut, and Rivet Manufacturers recently published a paper by J. O. Almen in which he reported some tests made on bolts. With a testing machine equipped with gravity scales, the load applied to a specimen could be accurately weighed at any position of the load applying device. The curves in Fig. 5 were drawn from data taken with such a machine and it appears that the reasonings thus far set forth are borne out with some amount of accuracy. Curve A was an ordinary stress-strain diagram and is simply an inclined straight line. The bolt was then arranged so that it could be tightened against relatively rigid abutments to an initial load of 4,000 lb. This initial load elongated the bolt 0.0046 inches. An external load was then applied to the bolt and the load extension curve B for those circumstances drawn. This new curve does not rise exactly vertical to join the straight load extension curve, as would occur if the bolts were perfectly rigid, but gradually curves to join it. This shows that the bolted members were elastic to a degree and that the load on the bolt was augmented by the elastic recovery of the bolted members.

The added load increment can be measured by drawing a line vertically from the 0.0046 point of the abscissa and intersecting the straight load extension curve. A line drawn horizontally over to the curve B will indicate the strain of the bolt under this load and if a vertical line is again drawn up to curve A, the load to which this strain is proportional can be found. The difference between this tightened load and the external load is called the "cyclic bolt load." From the method of analysis described above it is seen that the cyclic load on a bolt is increased as the elasticity of the bolted members is increased and will decrease as the elasticity of the bolt is increased.

The importance of the correct initial loading of bolts which fasten pieces with fluctuating load can be seen with these conceptions in mind. If there is no initial tightness of the bolt holding the bearing cap of a connecting rod the change

(Continued on page 28)

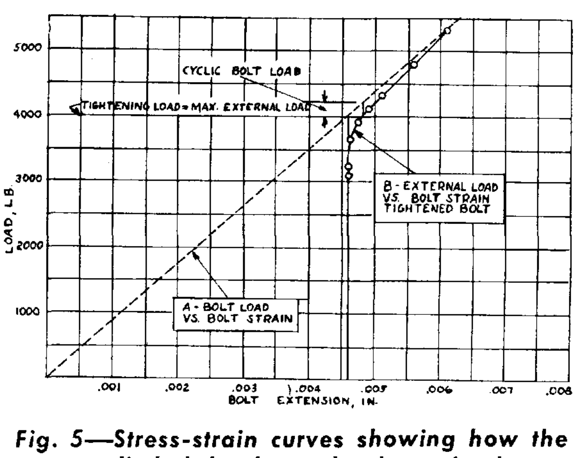

Fig. 5—Stress-strain curves showing how the cyclic bolt load may be determined

BOLTS AND BOLTED MEMBERS

(Continued from page 11)

in stress would alternate from zero, the initial load, to the maximum inertia load of the piston and connecting rod. Under this large stress change, the *fatigue* strength of the bolt would be less than one-fifth of its original static strength. If, on the other hand we tighten the nut with an initial tension load, there will be no change in stress, and therefore the operating stress is almost equal to the static strength. This is because the parts of the connecting rod that are bolted together are not as yielding as the bolt.

The arguments are also valuable in determining methods of increasing bolt life. A spring such as shown on the nut in Fig. 4 will increase the life of the bolt by adding to its elasticity. Likewise, any expedient that will increase the rigidity of the bolted members or increase the elasticity of the bolt are desirable and will increase the fatigue strength of the bolt.

The short studs used at times in engines cause an evil loss in bolt efficiency. If the thinness of the bolted member is such that any wearing away of its thickness will cause a decrease in the initial tightness of the bolt, fatigue failure will be the subsequent result. So it would appear to be a good idea to use long studs to reduce the amount of bolt tension lost when the thickness is worn away.